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DISPLACEMENT VENTILATION

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Synopsis

This paper presents information on the principles and application of displacement ventilation and discusses the most significant design parameters. Included are details of the performance characteristics of low velocity air terminal devices and of the resulting indoor environment based on experimental studies. Strategies for the design of displacement ventilation systems are also presented.

INTRODUCTION

The main purpose of ventilation is to provide good quality air for the occupants. Basically, this may be achieved in several ways. The more conventional method is to continuously dilute the indoor pollutants by mixing the room air with "fresh" incoming air and remove the polluted air from any suitable location. Another method is to introduce the "fresh" air at one part of the room and allow it to sweep in one direction across the space taking the pollutants with it and exhaust the polluted air at the opposite part of the room. This uni-directional method is often called piston flow, or plug flow. The flow is established by supplying air through a large opening in one room surface (often covering most of the surface area) and exhausting the air through a similarly large opening in the opposite surface. It is used in such applications as clean rooms and operating theatres. A more recently applied method called displacement ventilation has some characteristics similar to upward-moving piston flow but, unlike piston flow, it is largely dependent on heat sources in the room to provide the upward motion of the air.

In displacement ventilation, the incoming "fresh" air is gently introduced near the floor and at a temperature just slightly lower than the design room air temperature. The cool air flows slowly over the floor and air that has become warmed rises above the cool layer. Any local heat sources in the lower part of the room create convection currents that also contribute to the general upward movement and the air is ultimately exhausted at high level.

Although this type of system has been used in other countries, notably in Scandinavia, since 1970, there is little experience of its use in the UK. This paper is based on a BSRIA report[1] that contains state-of-the-art information on the application of this method of ventilation and data appropriate to its design and operation.

DESIGN CONCEPT

The objective in designing displacement ventilation systems is to provide slow moving "clean" air at a comfortable temperature around the occupants and retain the higher pollutant concentrations and temperatures above head level. Figure 1 illustrates this concept.

Air enters through a large terminal at low velocity and at a temperature slightly lower than the surrounding air temperature.

As a consequence, the incoming air tends to drop to floor level and continue to move slowly across the space. Within the room, local heat sources produce upward moving convective plumes into which surrounding air is entrained. The warm, often contaminated, plumes spread out below the ceiling where mixing takes place so that there is little difference in the temperature and the contaminant concentration between the air at the upper level and that at the extract.

The height of the boundary between the upper, mixed flow and the lower, un-mixed flow depends on the relationship between the incoming air flow rate and the rate of flow in the plumes. The boundary will stabilise at a level at which these two flow rates are equal (Figure 2). Below the boundary level, the upward rate of flow in the plumes is smaller than the supply air flow rate. As a result there is a general upward direction of movement in the lower part of the room and the temperature and contamination level of the air outside the plumes is close to that of the supply air.

However some increase in temperature with height above the floor is inevitable so care must be taken to keep this gradient within comfortable limits.

This type of ventilation is often described as buoyancy driven because of the major influence of the heat sources in creating the rising air flow pattern. The main advantage claimed for these systems is that clean and comfortable conditions can be maintained with air supplied at a lower flow rate and a higher temperature than that required to achieve an equivalent effect with the traditional "mixed flow" system. Thus significant energy savings can result.

KEY CHARACTERISTICS

Temperature Distribution

In displacement ventilation the slow-moving incoming air does not mix rapidly with the surrounding room air so the temperature of the air in the lower part of the room remains closer to the supply air temperature than to the exhaust air temperature (see Figure 3).

As Figure 3 also shows, the air temperature increases as the air rises through the main body of the room until it reaches the stratification boundary layer. There its temperature rises more steeply to a value close to that at which it is exhausted.

Air Velocities

Air is discharged at a velocity usually 0.5 m/s or less,into the room through supply air terminals located near floor level. Although the inlet velocity is low, and the temperature differential is small, a zone exists near the terminal in which conditions will not be acceptable for sedentary comfort. The extent of this zone, usually between 1 and 2 metres, depends on the type, size and shape of the terminal, its height above floor level and the supply air flow rate.

Further from the terminal, the air drifts across and up through the room so the velocities normally remain well below the upper limits for comfort.

The incoming air stream is not only characterised by low air velocities but also by low turbulence levels. This is an advantage because people are less sensitive to airflow at a given velocity when the turbulence is low than when it is high. Typically, the turbulence intensities associated with displacement ventilation are in the range 10% to 40%. However, comfort studies suggest that this advantage is offset by people's greater sensivity to cool air at ankle level[2].

Pollutant Concentration

The variation of airborne pollution in a room follows a similar general pattern to the variation of air temperature. Figure 4 shows a typical example of the increase in pollutant concentration with height. This illustrates the potential benefit of displacement ventilation in providing a cleaner environment for the occupants, as long as they are not directly affected by plumes arising from pollution sources located at low level.

Cooling Capacity

One of the most significant factors that limits the capability of displacement flow systems to remove heat in occupied rooms is the maximum tolerable vertical temperature gradient. On this basis, Sandberg[3] suggests that for office accommodation the application of displacement flow ventilation is limited to a room heat gain no greater than 25 W/m^2. Other authors suggest somewhat higher limits of 30 to 40 W/m^2 (e.g. Kegel[4]).

One way of increasing the cooling capacity of displacement flow systems is to recirculate some of the room air through an induction circuit. In this room air is induced into the supply air and is mixed before discharge through the low-velocity air terminal device into the room. This reduces the room air temperature gradient for a given heat load and hence allows larger heat gains, perhaps up to 50 W/m^2 to be dealt with.

For applications requiring greater cooling capacity, displacement ventilation may be used in conjunction with other cooling systems, for example, chilled ceilings.

The supply air temperatures used in displacement ventilation are generally higher than those in conventional systems so that advantages can be gained from smaller air-cooling plant and lower energy consumption. Indeed, if humidity control is not a primary requirement, displacement ventilation provides the potential in some applications for the elimination of mechanical cooling altogether.

As displacement ventilation is usually applied as a full fresh air system it is more often than not associated with heat recovery from the exhaust air so that energy economy can be achieved.

APPLICATION

Displacement ventilation has been applied to a variety of building types. Many of the early examples were industrial applications but the range has increased to include laundries, department stores, theatres, cinemas, restaurants and, most recently, offices.

Conditions under which displacement ventilation is most effective are as follows:-

- where the supply air is cooler than the room air
- where contaminants are warm and/or lighter than the surrounding air and associated with a heat source.
- where the surface temperatures of heat sources are high, e.g. greater than 35°C
- in tall spaces, e.g. where ceiling heights are greater than 3 m
- where disturbances to room air flows are weak

Conversely, displacement ventilation is least effective where:-

- the supply air is warmer than the room air

- contaminants are cold and/or more dense than the surrounding air
- surface temperatures of heat sources are low, e.g. less than 35°C
- ceiling heights are low, e.g. less than 2.5 m
- disturbances to room air flows are strong.

These features mean that displacement ventilation is particularly suited to industrial applications where cooling requirements are predominant, where heat sources and contaminants are at high temperature, and where ceiling heights are large. For example, Figure 5 shows the vertical temperature and contaminant concentration profiles for a process plant in a 20 m tall building. It illustrates the effectiveness of the ventilation system in retaining high temperatures and pollution levels at the upper part of the space. Applications in other types of building with similar characteristics can be equally effective.

In buildings with low ceiling heights effective application is more difficult. Hence, the limited use to date in office accommodation. The difficulties lie in ensuring that the stratification boundary layer is above head height and that the temperature gradients within the occupied zone are acceptable (Figure 6). Problems may also occur if contaminants are released at low level and entrained into the plume around a person so that unacceptably high concentrations are experienced at the breathing zone. The loss of usable space close to the low-level supply terminals is also recognised as a problem in occupied rooms.

As it is relatively easy to disturb the displacement flow pattern in a room, its application is not appropriate in areas where strong internal movements are expected nor in leaky, poorly insulated buildings where infiltration and down-draughts adversely affect the ventilation performance. In particular, down-draughts from cold surfaces can result in polluted air from high level being drawn downwards so causing increased levels of contamination in the occupied zone and thus degrading the effectiveness of the ventilation.

As indicated above, the displacement effect is lost when air warmer than room air is supplied. To meet heating requirements, separate systems such as those incorporating radiators, radiant panels or convectors are recommended.

SUPPLY AIR TERMINAL DEVICES

Physical Characteristics

Low velocity air terminal devices are very simple in principle. They consist of a plenum chamber into which supply air is ducted and from which air is discharged into the room through a large surface area.

Devices are available in many shapes, including:

* rectangular boxes with the outlet in one face (Figure 7)
* triangular plan boxes with the outlet in one face, for corner mounting (Figure 8)
* cylindrical, semi-cylindrical and quarter-cylindrical devices (Figures 9, 10 and 11)
* flat profile rectangular devices designed to spread the supply air (Figure 12)

Some form of medium is used at the outlet face of the terminals through which the air passes. Variants include perforated metal sheet, filter-like material and porous plastic foam. The free area ranges from about 10% to 50% or so.

Most devices are designed to discharge air at an even velocity over the whole area of the supply opening. While this could be achieved by the use of a very large plenum, most devices have small plenums that incorporate special features for equalising the discharge air flow. Alternative forms of these internal features are cones, orifices and nozzles, and angled perforated sheets.

Airflow Dynamics

As the air leaves the air terminal device at low velocity and at a temperature slightly lower than the surrounding room air, it flows at an angle downwards towards the floor. Under the influence of buoyancy forces this downward moving air tends to accelerate so that at some distance from the terminal the air velocities near the floor may be higher than the supply velocity. This phenomenon is sometimes termed the "cascade" effect (see Figure 13) and its significance depends mainly on the height of the top of the supply terminal and on the temperature differential. However, it has not been possible to adequately take these factors into account to enable the generalisation of the characteristics of flow at different conditions from low velocity air supply terminals of different types.

Further from the terminal, the air spreads across the floor in a shallow stream at reducing velocity. If undisturbed, the layer of air moving close to the floor remains stable and on reaching a solid wall will reverse its direction and return in contraflow just above the initial air stream. As these air streams move slowly across the room they feed convection currents generated by surfaces at a higher temperature so that, as already described, a general upward movement is created in the main body of the room.

The air stream issuing from the terminal is not only characterised by low velocity, but also by low turbulence. This is illustrated in Figure 14 which shows lines of constant turbulence intensity along a centre line cross-section taken normal to the face of the low velocity supply terminal. Comparing this with Figure 13 suggests that mean velocity and turbulence intensity are inversely related. Thus as the airstream progresses across the floor it becomes more turbulent.

Temperature Distribution

Research evidence to date indicates that the increase in the temperature of the supply airstream as it progresses across the room is not easy to predict. It seems to vary with supply air flow rate, type and height of the supply air terminal, and the difference between supply air and room air temperatures. With this uncertainty, the use of supply air temperatures more than a degree centigrade or two below the comfort temperature should be avoided where occupants are expected to be seated within a distance of 2 metres from the supply air terminal device.

DESIGN STRATEGY

The design choices are more complex in displacement flow systems than they are for mixed flow because of the increased number of factors. The factors that need consideration are:

- mean room air temperature
- temperature gradient
- supply air temperature
- overall temperature difference

heat loads
- overall
- at low level

supply air flow rate
room air contamination
selection of air terminal device

Covering all of these factors in one design strategy is difficult so it has become practice to focus the initial attention on either thermal comfort or indoor air quality, depending on which is the more critical. Both of these approaches are covered in the design procedures published by several manufacturers. In the absence of manufacturers' data, the following general procedures may be used. The emphasis in one is on thermal comfort and the other, air quality. The basis of each is illustrated in Figures 15 and 16 and given in the following sections.

Thermal Comfort Based Design

The starting point for this method is the selection of a temperature gradient limit, above which discomfort is likely to occur. The appropriate limit for sedentary occupation is 2 K/m, although some sources recommend a lower value of 1.5 K/m. For a standing person engaged in heavy work the limit can be extended to 2.5 or 3 K/m.

From the limited test data available, the relationship between the temperature gradient in the occupied zone and the overall temperature difference may be generalised as shown on Figure 17. The basic assumptions are that the temperature gradient is linear from a point 0.1 m above floor level to the ceiling, that the exhaust air temperature is the same as that at the ceiling, and that the difference between the temperature of the air at 0.1 m above the floor and the supply air temperature is 0.4 times the overall exhaust-to-supply air temperature difference. This ratio is in the centre of the range found in the technical literature, from 0.3 to 0.5.

These assumptions are considered valid for rooms with ceiling heights up to 4 m so they should cover most applications in which human comfort for sedentary occupation is the primary criteria.

From Figure 17, the following relationship can be deduced:

$$t_e - t_s = t_g \, (h_r - 0.1)/0.6 \qquad (1)$$

where t_e = exhaust air temperature (°C)
t_s = supply air temperature (°C)
t_g = air temperature gradient (K/m)
h_r = floor to ceiling height (m)

Using this equation, the overall temperature difference, $(t_e - t_s)$, can be calculated.

From knowledge of the room heat load, the required air flow rate can be calculated from the equation:

$$W = \rho C_p \, Q \, (t_e - t_s) \qquad (2)$$

where W = total estimated heat load (W)
ρ = air density (kg/m³)
C_p = specific heat of air (kJ/kg K)
Q = air volume flow rate (litres/s)

At normal room conditions, ρ = 1.2 kg/m³ and C_p = 1.02 kJ/kg K, so the equation to calculate Q reduces to:

$$Q = W/1.22\ (t_e - t_s) \quad (l/s) \qquad (3)$$

The supply air temperature may be selected from the following:

sedentary occupation	18-20°C
more active occupation	17-19°C
industrial applications	15-17°C

Air Quality Based Design

The starting point for this design process is the calculation of the total flow rate in the plumes rising above the various heat sources in the room when those plumes reach the upper boundary of the "clean" zone. The heat output or surface temperature of each source is needed together with their size and height above floor level.

The equations used for these calculations are given in Appendix A and, using these relationships, the flow rates at different distances above heat sources may be calculated.

Any unavoidable downward convective flows should be subtracted from the plume flow rate total to give the required supply air flow rate.

From this and the overall heat load (taking into account the loads both below and above the clean zone boundary), the overall temperature difference may be calculated from equation 3.

Using equation 1, a check can be made on the temperature gradient in the room expected at the derived overall temperature difference for the given room height.

The supply air temperature may be selected from the values given in the previous section.

Selection of Low Velocity Air Terminal Devices

Having determined the required air flow rate and supply air temperature, the next stage is to select suitable low velocity air terminal devices. To achieve uniform air distribution, throughout a space, it is generally better to use several smaller devices rather than one large unit but such factors as the room layout, seating arrangements and space constraints will affect the choice of suitable locations.

At present, it is not possible to generalise the performance characteristics of low velocity terminals, so to make selections it will be necessary to refer to manufacturers' literature. In general, to keep the "near zone" (that is, the zone close to the terminal where the airstream is likely to cause discomfort) as small as possible, it is best to select a device with a large free area, low discharge velocity, a wide spread and the lowest possible height.

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APPENDIX A

Equations for the determination of the air flow rate in plumes have been presented by Skaret[5]. From these, the following relationships have been derived assuming a room air temperature of 22°C.

For point heat sources -

$$Q = 0.006\ w^{1/3}\ (x + x_p)^{5/3}\ (m^3/s).$$

For linear heat sources -

$$Q' = 0.014\ w'^{1/3}\ (x + x_p)\ (m^3/s \text{ per } m).$$

For vertical heated surfaces -

$$Q' = 0.0028\ t^{0.4}\ z^{1.2}\ (m^3/s \text{ per m width}).$$

where:

Q = air flow in plume at distance x from source.

Q' = air flow per m length in plume at distance x from source.

w = rate of heat release (W).

w' = rate of heat release per m length (W/m).

x = vertical distance from upper surface of source (m).

x_p = vertical distance from apparent origin to upper surface of source (assumed equal to the diameter or width of the heat source) (m).

z = height of heated surface (m).

t = temperature difference between surface and surrounding air (K).

These equations are based on the assumption that there is no vertical temperature gradient in the surrounding air. Whereas Skaret claims that for normal applications this may be ignored, Danielsson[6] argues that the convective flow can be substantially reduced even with gradients as low as 1 K/m. Further research[7,8,9] is being undertaken on this subject but until further evidence becomes available it is recommended that the full convective flow rates are used in design to ensure that the stratified boundary is at, or if not above, the required level.

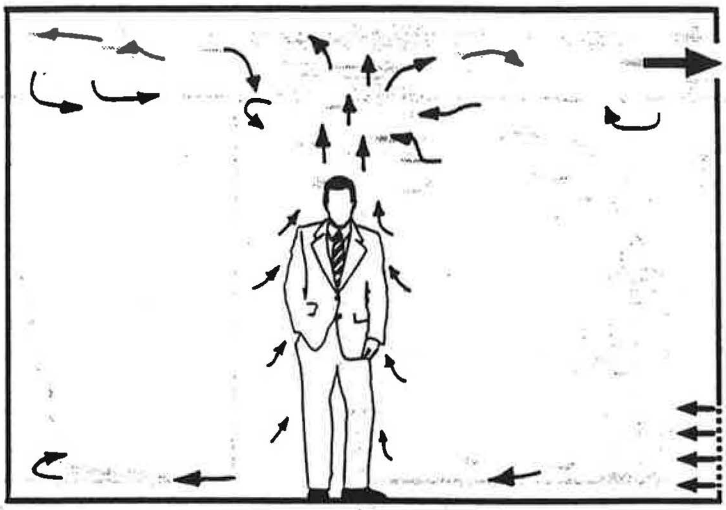

Figure 1 Buoyancy driven displacement flow

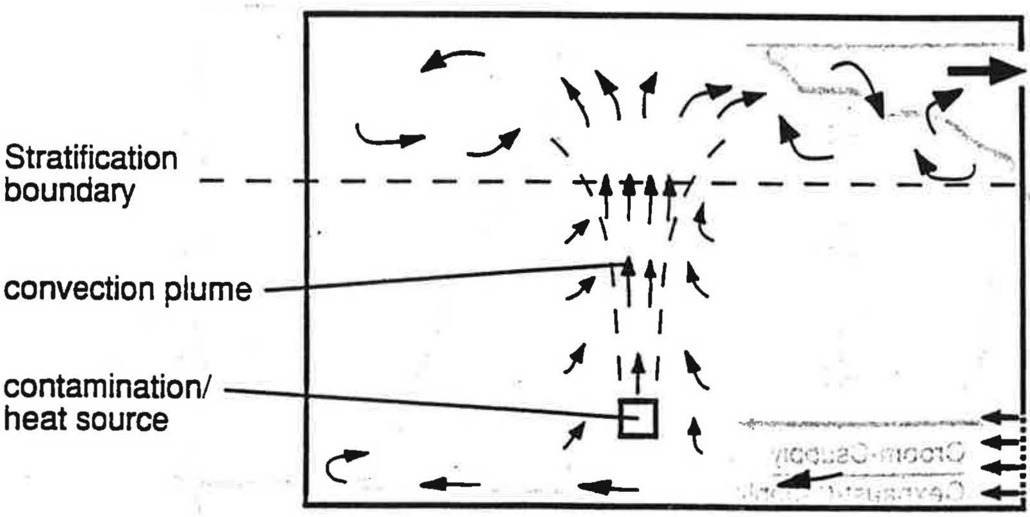

Figure 2 Stratification boundary

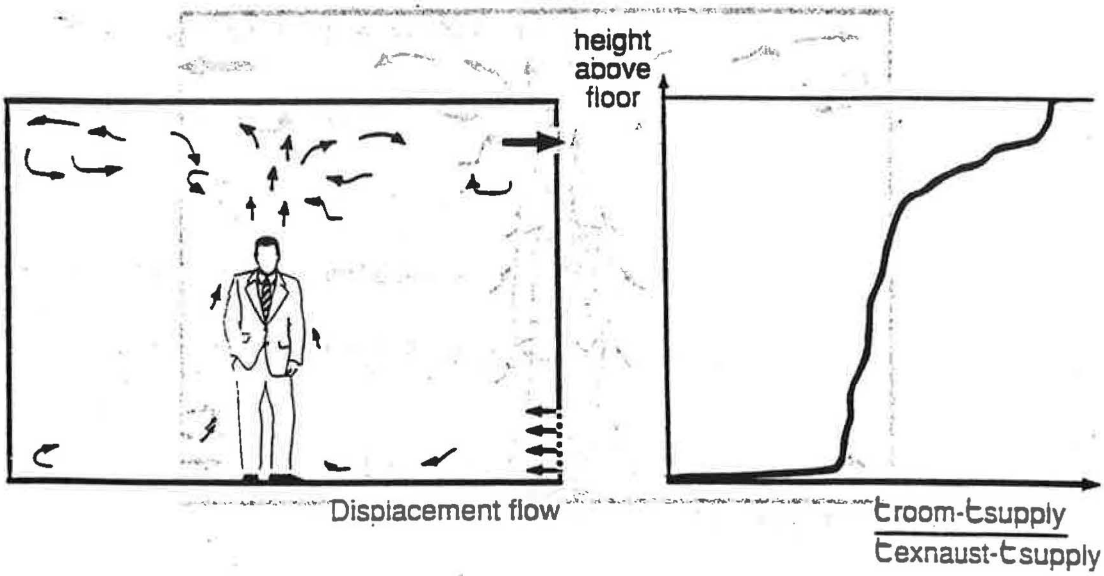

Figure 3 Typical temperature profile

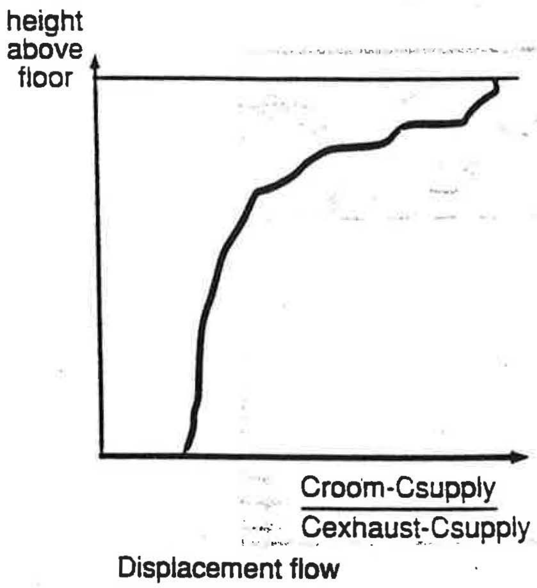

Figure 4 Contaminant concentration profile

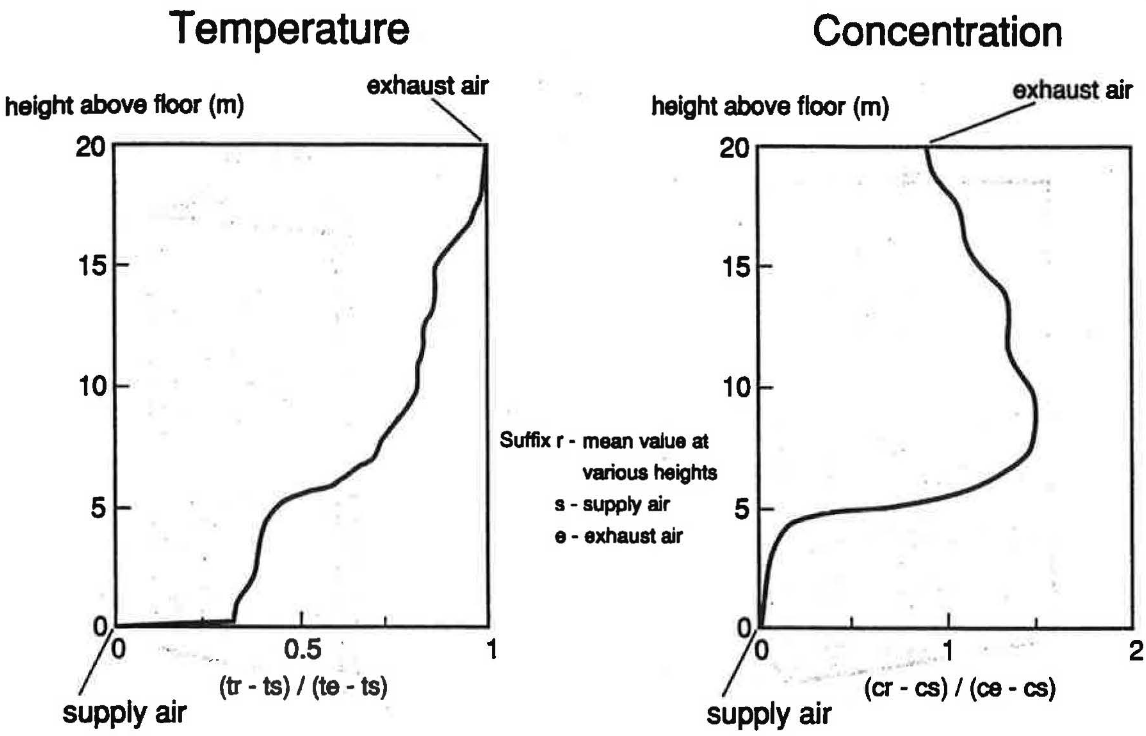

Figure 5 Temperature and concentration profiles in process plant

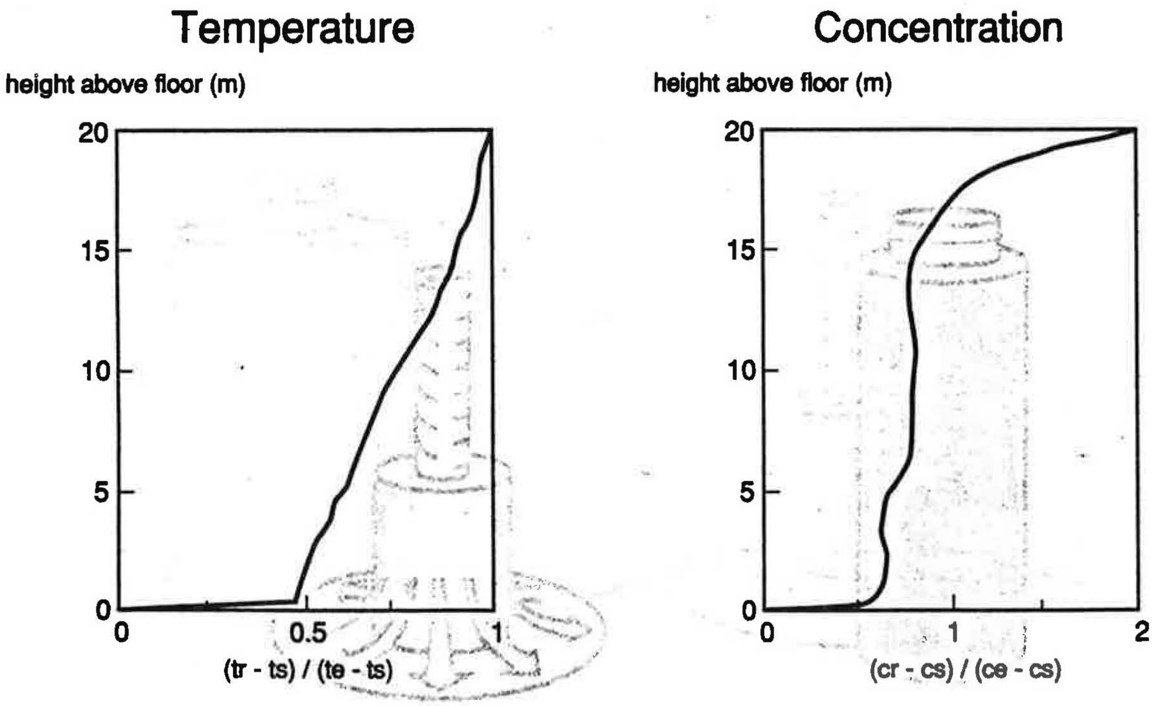

Figure 6 Displacement ventilation in an office

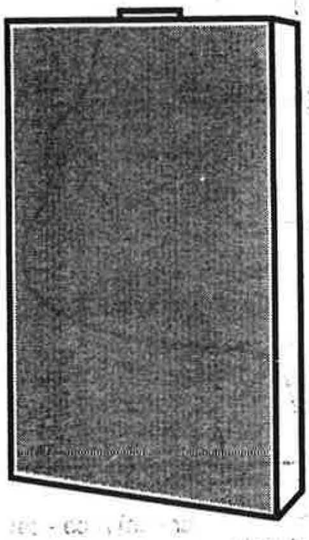

Figure 7 Rectangular low velocity air terminal device

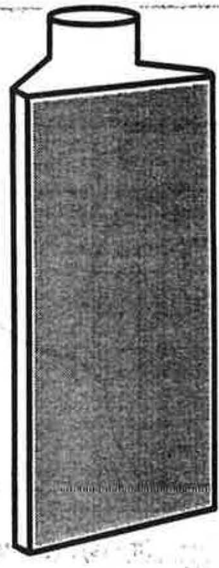

Figure 8 Triangular plan low velocity air terminal device

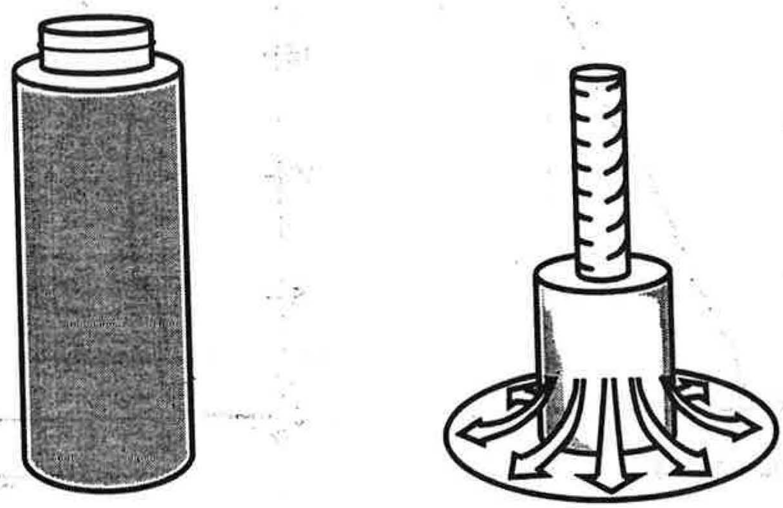

Figure 9 Cylindrical low velocity air terminal device

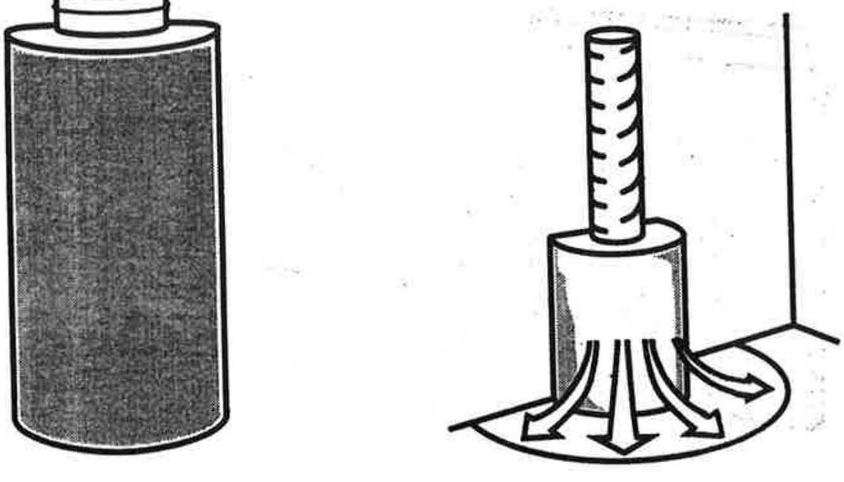

Figure 10 Semi-cylindrical low velocity air terminal device

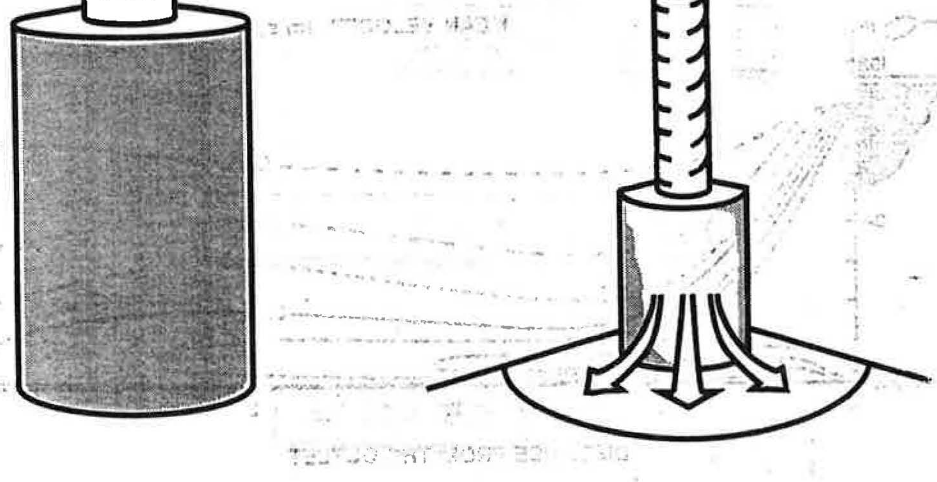

Figure 11 Quarter-cylindrical low velocity air terminal device

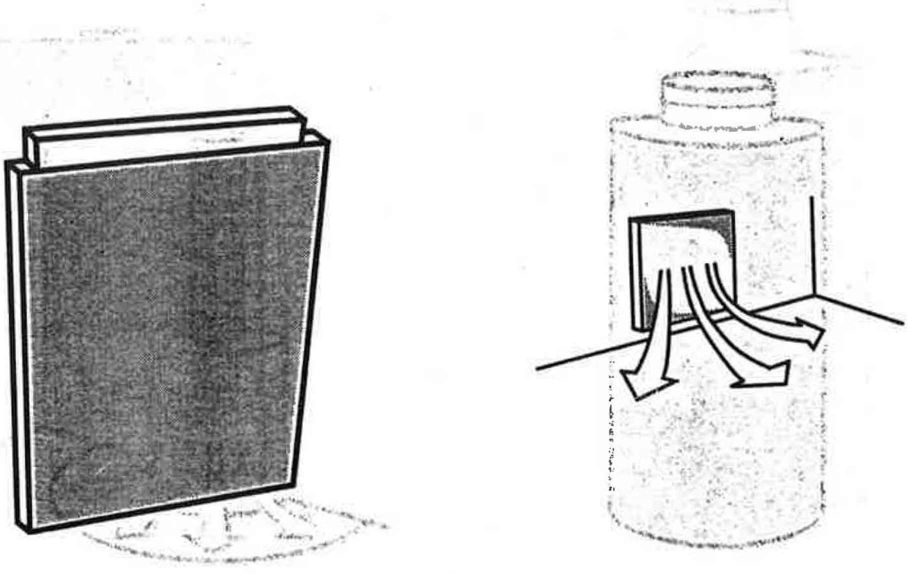

Figure 12 Flat-profile low velocity air terminal device

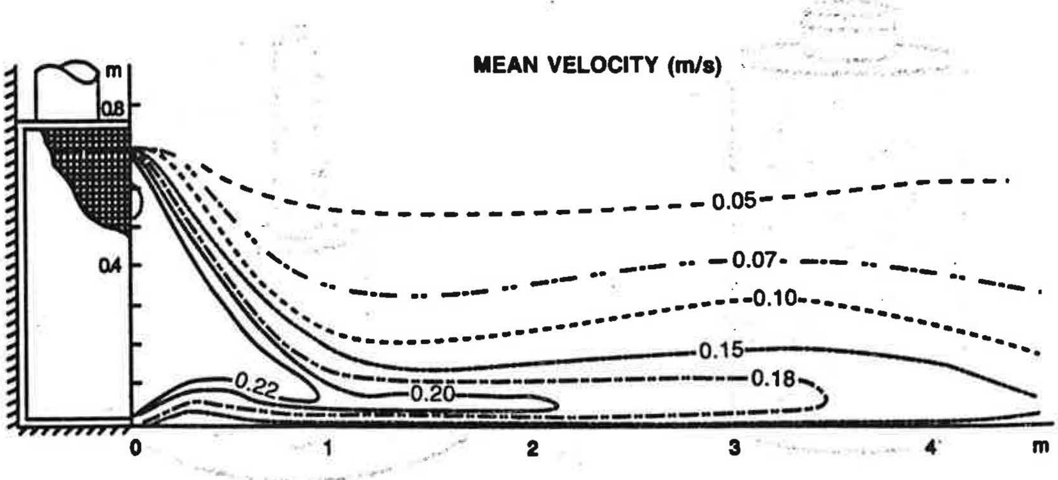

Figure 13 Velocity characteristics near air terminal device

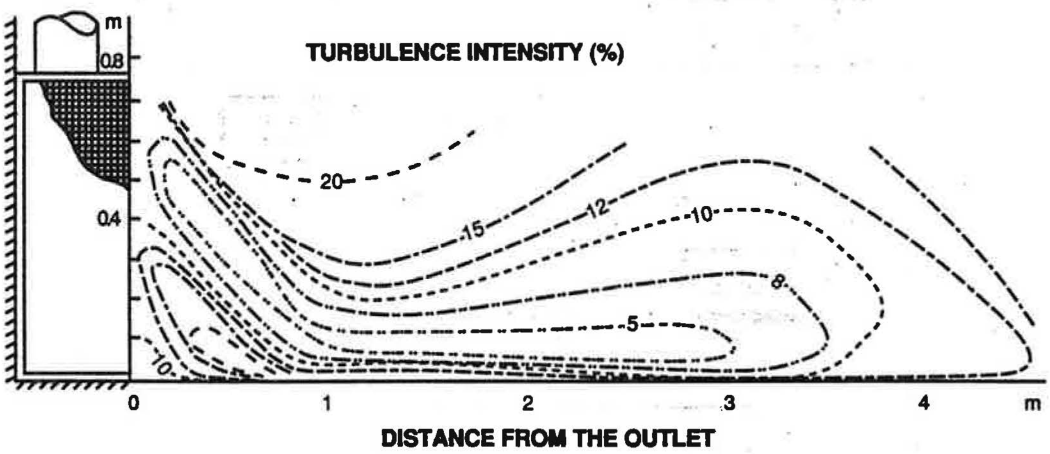

Figure 14 Turbulence characteristics near air terminal device

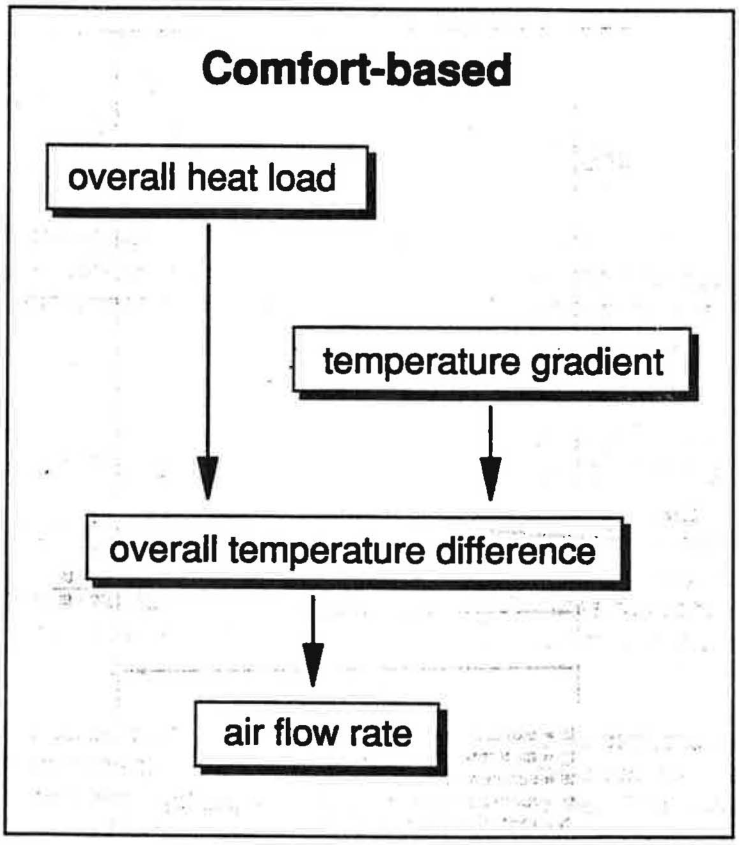

Figure 15 Comfort based design strategy